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CONCEPTION OF
WHEEL/RAIL SYSTEM TEST RIG

DSL No: 227-1

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The contents of this report reflect the views of the contractor and not necessarily the official views or policies of the Transportation Development Centre of Transport Canada

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17. Abstract

A preliminary investigation into the conceptual definition of a wheel/rail system test rig was carried out. The investigation tasks included a literature search and an assessment of potential practical applications. Following the consideration of the factors important in the physical simulation, specifications were drawn up for a wheel/rail system test rig. A design concept was prepared taking into account the functional and technical specifications. The investigation concluded that a full-scale multipurpose wheel/rail system test rig is not only technically feasible but highly desirable as a cost-effective research and development tool. It is recommended that detailed design definition be carried out on major subsystems of the test rig.

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17. Résumé

Une recherche préliminaire a été faite en vue de la définition d'un banc d'essai pour étudier le comportement dynamique du système roue/rail. Elle a commencé par une recherche documentaire et une évaluation des possibilités d'utilisation. Après la détermination des facteurs physiques en jeu, les caractéristiques et le comportement attendus du banc d'essai ont été couchés sur papier, puis un concept a été imaginé en conséquence. Le rapport conclut qu'un banc d'essai polyvalent en vraie grandeur serait non seulement faisable techniquement, mais aussi qu'il constituerait un outil économique de recherche et de développement. Il recommande qu'une définition détaillée soit faite des principaux composants du banc.

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SUMMARY

A preliminary investigation into the conceptual definition of a wheel/rail system test rig was carried out. The investigation tasks included a literature search and an assessment of potential practical applications. Following the consideration of the factors important in the physical simulation, specifications were drawn up for a wheel/rail system test rig. A design concept was prepared, taking into account the functional and technical specifications.

The investigation concluded that a full scale multipurpose wheel/rail system test rig is not only technically feasible, but highly desirable as a cost-effective research and development tool. It is recommended that detailed design definition be carried out on major subsystems of the test rig.

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1. INTRODUCTION

Further to a meeting organized by The Transportation Development Centre (TDC), with representatives of CN Rail, CP Ltd., and the National Research Council (also attended by the author), DSL was awarded a contract to pursue work towards the conceptual design of a wheel/rail system test rig.

The study objective was to define a preliminary concept of a full-scale multipurpose wheel/rail system test rig capable of supporting investigations on wheel-rail interaction, brake and bearing problems.

The report contains information gathered in a literature search, the result of application assessment study, test rig specifications and definition of a preliminary concept.

Conclusions drawn from the investigation, as well as recommendations for further studies, are also included.

2. DESCRIPTION OF ACTIVITIES

Project tasks carried out in this phase of the project are described below:

a) A collection of technical material relevant to the proposed system was made. Part of the material was made available by Mr. J. Kalousek of the National Research Council. Other references were gathered from various sources during the project. A list of the material collected is included in Appendix A.

b) A review was made of a list of potential practical applications; this list had been initially prepared by the project steering committee members. This review, in conjunction with the technical material collected, formed the basis for an assessment of the various factors that needed to be addressed in a wheel/rail system test rig. It was not within project scope to provide a detailed critical assessment of potential applications.

c) From considerations on the various factors found important in the physical simulation of a wheel/rail system, specifications were drawn up. A copy of the specifications is included in Appendix B.

d) A design concept with the potential to meet the specifications was prepared. A description of the concept is included in Section 3.

e) The project also involved attendance at a symposium on experimental and theoretical works in wheel rail interaction phenomena (see Reference 27 in Appendix A). This allowed the author to obtain an overview of recent developments in this field of activity, as well as to provide an opportunity to personally contact people with expertise in specialized areas.

3. DEFINITION OF A CONCEPT

A concept which is believed to have potential in meeting the specifications is described. A short discussion is also provided concerning a few options worthwhile highlighting at this stage.

3.1 Fundamental Features

The proposed concept is summarized below.

a) The rail simulation device is a conventional wheelset assembly having the rims machined as rail heads. The drive motor is located inboard of the discs. The wheelset and motor are mounted within a standardized subframe which contains all internal reaction forces (i.e. journal forces, and motor reaction force).

b) The test specimens are full-size wheelset assemblies, also mounted within a standardized subframe, which contains internal reaction forces originating from the brakes and the journal bearings (Reference 19, Appendix A, describes an example case of standardized subframe).

c) The two wheelset specimens are articulated to one another in order to obtain the required relative positions and reaction forces for test. Both subframes are mounted on a rigid main frame which is sprung from the ground. In addition to movability, this configuration allows the floor design requirements to be adequate for static weight only, as opposed to impacts and vibrations resulting from tests (e.g. eccentricity of the wheels, or tread surface defects).

d) For tests requiring controlled longitudinal slip at the wheel-rail interface, the wheelset specimen is fitted with a motor similar to the rail simulation device. The articulation between the wheelset specimen allows to separate two of the four discs in contact. The desired creepage can then be set on the remaining wheels in contact. DC motors allow to connect into a motoring regenerative braking configuration so that only the energy dissipated needs to be supplied.

3.2 Discussion

a) Rail Simulation Device

The conventional locomotive wheelset-motor assembly appears attractive, to meet requirements of strength to withstand shock forces from tread surface defects. Actual torque and power capabilities may be close to the specified requirements, at least for short time durations. Also, interesting options exist, such as the use of replaceable rims for wear tests, or the use of existing resilient wheels to reproduce track elasticity.

b) Actuators Configuration

From a conceptual point of view, there is a large number of possible configurations, and more than one configuration deserves to be examined at the design stage. The considerations below illustrate this point.

There are five independent relative motions that need to be addressed (neglecting deformations of the wheelset assemblies themselves).

- Vertical and roll (i.e. with respect to common railway terminology), for the control of wheel load, and for the control of the angle between the wheel and the rail profile (when this proposed feature is used, as explained in 3.1"d").

- Lateral and yaw, to control the tracking error (or the lateral force), and the angle of attack.

- Longitudinal, for the control of slip at the wheel rail interface.

The longitudinal control can only be provided by motor control (or brakes). The other four controls, however, can be done in various ways depending on whether articulation is provided between the two frames, or between the frames and the wheelsets. The complete rig assembly can also be configured vertically (wheelset specimens above the rail simulation device), or horizontally. Finally, several types of actuators can be used, as well as several types of articulations or connections. Generally, a specific configuration should be designed from consideration of the various logistic aspects of running experiments. Considerations should also be given to safety and dynamic stability, especially if the rig is to be implemented on a movable platform.

c) Longitudinal Creep Control

The requirement for high resolution (i.e. fractional percentage) control of longitudinal creepage is not believed to be a technical problem in itself, for the following reasons. For test cases where a small creepage is desired, it is possible to use motor torque as a control variable. In this range of operation, creep force is a sensitive variable of creepage. The evaluation of creepage can be done by the use of angular position encoding transducers (i.e. toothed discs, or shaft encoders). Two such transducers arranged in a differential mode can provide, over a finite elapsed time, very high resolution with respect to cumulative creepage distance and therefore with respect to mean creep velocity.

At high values of creepage however, the creep force would not be a suitable variable to control the motors, and therefore speed would be used as a control variable. At high creepage, however, the fractional percentage accuracy is not required.

From these considerations, it is evident that different motor control strategies need to be present in the final control system; however, this problem has not been examined in detail.

4. CONCLUSIONS

a) It appears that a full-size, single wheelset test rig would provide a powerful tool to enable controlled condition testing on a wide variety of aspects relating to wheels, rails, brakes and bearings. Applications would cover aspects relating to many present practical problems, but perhaps more importantly, it would also give a tool and provide data to rationalize and improve on engineering design and analysis methods.

b) The test rigs reviewed in the literature search, generally have a limited number of controllable test conditions, and in this sense they can be classified as special purpose compared to the proposed one.

Judging from the various practical problem areas mentioned by the steering committee members, it appears that the proposed multi-purpose feature (especially the combination of controlled wheel-rail conditions and controlled brakes) is required, in order to reproduce conditions typical of actual service condition.

This point can also be illustrated by the following example. The reproduction of realistic wheel stress conditions requires controlled wheel-rail forces in three directions including contact stresses and heat generation at the wheel-rail interface, as well as brake forces and heat generation. It is often important to reproduce simultaneous combined condition since it is not always possible to superimpose effects of several factors taken separately.

c) Presently there is no standard technique to convert test results from small scale specimen to full-scale applications when dealing with many aspects of wheel-rail systems interaction. The difficulty of conversion is multiplied when dealing with coupled effects such as thermal, lubrication, wear and certain aspects of fatigue.

d) It is generally accepted by experts that full scale controlled tests would be useful to bridge the gap between small scale tests (or theories) and practice. This approach would minimize the risks and costs involved with in-service testing, and further, provide test data to compare with the analytical results.

e) It is considered technically feasible, and probably cost effective in comparison with other methods of obtaining comparable data, to implement a test rig capable of performing a variety of tests on single, standard full-size wheelset assemblies.

5. RECOMMENDATIONS

a) The work already started with the project subcommittee (i.e. application to present practical problem areas) should be pursued in more detail. In addition, it should consider applications where the generation of basic experimental data would help develop more systematic engineering analysis and design methods. This is important in order to highlight all important functional requirements for the proposed rig.

b) Design activities should be initiated to assess the practical feasibility of meeting the functional requirements.

c) Considering that no such multi-purpose test rig is yet known to exist, it is recommended that the early experimental activities be conducted in parallel with design activities. This could include for example, the setting up of a rig having limited capabilities compared to the proposed one. Use of available standard railway components would reduce costs, accelerate familiarization, and highlight at an early stage potential difficulties or perhaps simple solutions to specific test rig design problems.

APPENDIX A

LIST OF RELEVANT REFERENCES

This appendix contains a list of documents found to be relevant to the design of a wheel-rail system test rig.

1. "The Rail Dynamics Laboratory: Capabilities/Tests", Transportation Test Center, Pueblo, Colorado.

2. Kalousek, J., Bethune, A., "Rail Wear Under Heavy Traffic Conditions". Rail Steels-Development, Processing, and Use, ASTM STP 644. D.H. Stone and G.G. Knupp, Eds. American Society for Testing and Materials, 1978, pp. 63-79.

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17. Krause, H., Senuma, T., "Investigation into the Influence of Dynamic Forces on the Tribological Behavior of Bodies in Rolling/Sliding Contact with Particular Regard to Surface Corrugations", ASME Journal of Lubrication Technology, January 1981, Vol. 103/26-34.

18. Egelkraut, K., "Computer Controlled Test Stand for Wheel Set Bearings of the Federal German Railways - Conception and Operating Experience", Hanover Fair Seminar, 23 April, 1980.

19. "Lucas Girling Commissions 1 million Pounds (sterling) Brake Dynamometer", Railway Gazette International, June 1982.

20. "Manually Operated Tribometer", Railway Gazette International, June, 1982.

21. Kumar, S., Prasanna Rao, D.L., Rajkumar, B.R., "Angle of Attack and Wheel-Rail Wear", December 1981, prepared for U.S. Dept. of Transportation, FRA.

22. "Railroad Freight Car Hot-Boxes", Final Report, September 30, 1954, Armour Research Foundation of Illinois Institute of Technology.

23. Kalousek, J., "Rail/Wheel Wear Testing Facility", National Research Council Canada, Volume 13, No. 1, July 1981.

24. Krause, H., Senuma, T., "A Contribution Towards Improving the Applicability of Laboratory Wear Tests to Practice", paper presented at the International Conference on Wear of Materials, 1981, San Francisco, March 30 April 1, 1981.

25. Brockley, C.A., "Tribology in Canada", The Activities of the Associate Committee on Tribology, Engineering Digest.

26. "Metropolitan Railways to Become Less Noisy", BMFT Newsletter, February 1980.

27. Preliminary Proceedings of the International Symposium on Contact Mechanics and Wear of Wheel-Rail Systems, Vancouver, July, 1982. (The Preliminary Proceedings consist of 25 papers relevant to various experimental and theoretical aspects of wheel-rail interaction phenomena).

28. Gregory, D., McGinness, D., Prescott, N., "Railroad Wheel Failure Mechanisms and Test Facility Project/ Test Facility Design", June 15, 1981. Prepared for U.S. Department of Transportation by Jet Propulsion Laboratory, California Institute of Technology, through an agreement with National Aeronautics and Space Administration. Report No. 5030-502.

APPENDIX B

FUNCTIONAL AND TECHNICAL SPECIFICATIONS

The following pages outline specifications for a test rig intended to carry out realistic controlled condition tests relating to the interaction between the wheel, the rail, the brakes and the bearings.

The specifications are considered tentative at this time, since it has not been possible to review its content in detail with experienced people in the various fields of application.

To provide some perspective with regard to the requirements of various applications, the specification items have been classified as follows:

G - General

W - Wheel Rail Wear Tests

B - Brake Tests

J - Journal Bearing Tests

M - Miscellaneous

TERMINOLOGY

1. - TEST RIG: means the test rig facility in a general sense.

2. - RAIL SIMULATION DEVICE: means in a general sense, the device(s) used to simulate the rail contact phenomena.

3. - COORDINATE SYSTEM: when describing the various motions and forces on a wheelset assembly, the following notation is used:

   - Longitudinal indicates a direction corresponding to the normal motion of the simulated vehicle on the track.

   - Lateral indicates a direction transverse to the direction of motion and in the plane of the track.

   - Vertical indicates a direction along the normal gravity vector of the simulated railway vehicle.

   - Roll is a rotation about the longitudinal axis.

   - Pitch is a rotation about the lateral axis.

   - Yaw is a rotation about the vertical axis.

4. - Additional details are provided at the end for those items marked by a bracketed number.

- General

The following set of specifications covers general aspects of the rig. These include the type of equipment to be tested and any requirements that are of a general nature, irrespective of the type of test conducted.

G1 - The test rig must be able to accept, as complete assemblies, standard full-size wheelsets from freight cars, locomotives, passenger coaches and mass transit vehicles having standard gauge. The wheelset is meant to include auxiliary equipment such as brake discs, traction motor, and integral journal bearing parts, but not necessarily including the journal boxes.

G2 - The test rig must be able to accept, without permanent damage to hardware, actual brake system components that are in direct contact with the brake shoes (for tread brakes) or brake pads (for disc brakes). The types of railway brake equipment to be considered must include that now on conventional freight cars, freight locomotives and disc brake systems used on existing Canadian passenger equipment.

G3 - Peripheral equipment required to handle wheelset assemblies must be designed to a capacity of 6,804 kg (15,000 lb) which corresponds to the approximate weight of a conventional wheelset traction motor assembly.

G4 - The track simulation device must have a nominal static wheel load capacity of up to 22,680 kg (50,000 lb).

G5 - The test rig must be able to withstand in operation mechanical shocks and vibrations resulting from normal manufacturing and in-service tolerances. These include wheel eccentricity, out of balance, as well as possible wheel tread surface defects such as flat spots and shells up to allowable AAR limits.

G6 - If rollers are used to simulate track, the test rig must be complemented and supplied with means of assessing errors caused by the roller diameter effects as compared to actual track conditions. These errors consist of differences in contact areas, differences in contact pressures, and differences in wheel/rail contact geometry due to effects of wheelset yaw.

G7 - The rail simulation device must have means within itself to measure wheel rail forces on both wheels. Wheel rail forces must include as a minimum the axial components in the vertical, transverse and longitudinal direction, but not necessarily the moments about these directions[1].

- Wheel-Rail Wear Tests[2]

The following specifications cover additional items relating to wheel and rail wear studies or tests.

W1 - The test rig must be able to impose forces on the wheelset assembly having a magnitude of up to 6,804 kg (15,000 lb) in the lateral direction, in addition to the vertical load requirement of 22,680 kg (50,000 lb).

W2 - The test rig must have a force capacity to produce continuous slip or creepage in the longitudinal direction up to levels of Coulomb friction on the wheels of the test specimen wheelset. A Coulomb friction coefficient of 0.4 can be used at the full vertical load rating of the test rig (22,680 kg - 50,000 lb).

W3 - Corresponding to item number W2 (continuous slip), the test rig must have means of continuous heat dissipation up to a capacity of 100 kw, as well as means to control wheel and simulated rail temperatures, to less than 93°C (200°F) [3].

W4 - The test rig must have means to vary the wheel-rail contact geometry parameters resulting from the two following effects: i) relative roll motion (about the longitudinal axis[4], ii) relative yaw motion (about the vertical axis). The range of variation must be ± 3 degrees in both directions.

W5 - The test rig must have means to control longitudinal creepage at the wheel rail contact points up to values of 15 percent of the rolling velocity with an accuracy of .015 percent [5].

W6 - The test rig must have means to apply tread brakes during the wheel-rail wear tests. Tread brake actuators must have a normal force capacity of 4,536 kg (10,000 lb) and a transverse force capacity of 907 kg (2,000 lb). Means must be provided to measure the three axial force components and moments on the brake shoes.

W7 - The test rig must have means to produce during a test and under automatic control, slow cyclic variations (typically 1 cycle/minute) of the following factors: vertical force, lateral force, longitudinal creepage (or force), roll angle, yaw angle, and braking force.

Means used to produce the cyclic variations must be programmable so that arbitrary patterns of waveform, amplitude and phasing be selectable for test.

W8 - The test rig must have means to measure wheel and rail profiles without demounting the specimen. Equipment must be accurate to 0.0254mm (0.001") and repeatable to 0.00254mm (0.0001") [6].

W9 - The test rig must have means to monitor wear at the two standard gauging point of the wheel profile and simulated rail profile without interruption of wear test.

W10 - The test rig must be equipped with means to refresh and restore wheel and rail profiles without dismantling the wheelset from its mounting structure or frame. (Requirements for the surface finish and the machining methods need to be determined).

W11 - The test rig must be equipped with the necessary devices for automatic control and data collection, as well as for complete health monitoring (safety against failures), such that unattended operation is possible.

- Brake Tests

The following specifications cover additional items specific to brake tests.

B1 - The test rig must be configured such that the setup operations (e.g. installation and instrumenting of brake equipment) need not be performed on the rig itself. A quick procedure for mounting and demounting test specimens is desirable.

B2 - Power for the braking tests must be supplied via the rail simulation device.

B3 - The rail simulation device must have a power capacity for short time duration corresponding to 4,536 kg (10,000 lb) [7] longitudinal tread force at 201 km/hr (125 m.p.h.).

B4 - The rail simulation device must have means to execute under automatic control, pre-selected braking tests. In order to produce accurate results in stopping distances, speed profiles and other brake related variables, the control procedure must take into consideration the weight of the simulated vehicle, the track grade under consideration and wind resistance effects.

B5 - The test rig must have means to measure and record wheel tread and rim temperature dynamically during tests.

B6 - The rail simulation device must have means to control the temperature of the simulated rail to less than 93°C (200°F.).

B7 - The test rig must have means to simulate the cooling effects of wind during the braking tests.

- Journal Bearing Tests

The following specifications cover additional items relating to journal bearing tests.

J1 - The test rig must provide a mounting assembly able to accept a variety of journal roller bearings outer rings currently used on locomotives, passenger cars and mass transit vehicles. The stiffness characteristics of the mounting assembly must be nil in the roll and yaw directions.

J2 - The test rig must provide a mounting assembly able to accept a variety of standard journal roller bearings adaptors used on freight cars. The stiffness characteristics of the mounting assembly must be nil in the roll and yaw directions.

J3 - The test rig must be able to accept a variety of journal friction bearing assemblies. (Types of bearings and method of interfacing with bearing elements to be determined).

J4 - For roller bearings, the test rig must provide means to impose selectable moments about the roll and yaw axis, up to a value of 27,116 N.m (20,000 ft-lb).

J5 - The test rig must have means to perform controlled tests on bearings concerning running speeds, vertical and lateral loads, lubrication parameters, temperature distribution, and energy dissipation.

- MISCELLANEOUS

Dynamics Tests (noise, rail corrugations, unsprung weight vibrations).

M1 - The rail simulation device must have lateral and vertical stiffness characteristics whose values are within the range of values present on common North American railway track structure.

Derailment Tests (flange climbing)

M2 - In addition to the requirements outlined for the wear tests, the test rig must have a lateral force capacity up to 18,144 kg (40,000 lb).

M3 - The test rig must be able to execute under automatic control, pre-selected test procedures regarding flange climbing mechanisms. Requirements for the test factors needing control are the same as those outlined in item W7.

M4 - The test rig must be configured for fully safe recoveries from simulated flange climbing actions.

Climatic Factors Tests

M5 - In addition to requirements outlined in the various types of tests, climatic conditions tests require a control of temperature humidity and wind effects.

Hydroplaning of Brakes

M6 - The test rig must have means to supply and control variable quantities of water to the rail simulation device.

Track Lubricators

M7 - The test rig must have means to supply and control variable quantities of lubricants to the rail simulation device.

NOTES

(1) The measurement of moments could be claimed useful, however this is considered as a technical difficulty and may not be essential. The same consideration would apply for means of measuring separately the flange force and the tread force (applicable when two points of contact are present).

(2) Some of the specifications relating to wear may appear excessive. The material presented in Appendix C should help in clarifying some of these issues.

(3) Estimated from personal experience (qualitative only) relating to the running of heavy trains on curved track.

(4) Estimated from considerations of elasticities present on conventional North American track structure.

(5) The accuracy requirement is selected arbitrarily. For illustrative purposes, 0.015 percent represents approximately:

i) 30% of the creepage resulting from typical assembly tolerances regarding the matching of new wheels on a common axle.
ii) The creepage nominally present in the negotiation of a 0.3 degree curve.
iii) The creepage required (by calculation) to produce 5% of the coulomb friction force on a wheel loaded to 13,608 kg (30,000 lb).

(6) The accuracy value has been set arbitrarily, considering that a high accuracy in measurements will give the possibility to reduce the duration of wear tests.

(7) A lower level of force would suffice if this type of test were required only on lighter weight passenger coaches.

APPENDIX C

Computer Simulation Data expressing typical functional requirements for a wheel-rail wear test.

Example case study of a conventional 100-ton freight car truck.

INTRODUCTION

For illustrative purposes, an example case study is presented below, representing external factors acting on the wheels of a two-axle freight car truck during steady state curve negotiation. Parameters for the study were based on best estimates for the characteristics of a conventional North American freight car truck on standard track gauge. Wheel rail geometry are representative of the standard AAR Wide Flange Profile on a 254 mm (10") radius crowned rail head. Under this geometry there are two points of contact between the wheel and the rail.

A more detailed description of the simulation model used can be found in TDC Report No. 3211 - "Comparative Curving Analysis of Freight Car Trucks".

LIST OF FIGURES

The figures express a selection of variables as a function of track curvature.

1 Lateral Tracking position:
Positive values represent motion towards the outer rail. The flangeway clearance is assumed to be 12.5mm (½ inch).

2 Yaw Angle:
Angle of attack of the wheelset with respect to a curve radial line. Positive values indicate that rolling direction is towards the outer rail.

3 Net Lateral Force on the Wheel:
Positive indicates a force on the wheel applied towards the track centerline.

4 Wheel Slip:
Relative angular slip velocity of the wheel disk with respect to the linear velocity of the respective rail. Positive slip indicates slipping in the direction of motion (equivalent to braking).

5 Wheel Torque:
Resulting torque on the wheel disk about axle centerline. Positive values indicate that the wheel is producing a tractive effort in the direction of motion.

6 Rolling Resistance:
Represents the energy dissipation at the wheel-rail interface due to Coulomb friction. Requirements for power dissipation can be obtained based on the operating speed as shown in the figure.

7 Tread Creepage Magnitude in percent of the vehicle running speed.

8 Tread Creepage Direction:
Zero degrees indicated creepage in the direction of motion. 90° indicate creepage towards the inside of the curve.

9 Flange Contact Point Advance:
Under conditions of wheelset yaw angle, the point of flange contact moves forward with respect to the wheel position.

10 Magnitude of the Flange Force:
This force is normal to the flange face which is assumed at 70 degrees from the horizontal plane. The magnitude of the tangential flange force (which is not plotted) can be calculated using a Coulomb friction coefficient of 0.3 (estimated).

11 Magnitude of the creepage on the flange, in terms of percentage of the vehicle speed.

12 Direction of the creepage vector of the flange with respect to the rail. Zero degrees represents creepage in the direction of motion, 90 degrees indicates that the creepage vector is normal to the direction of motion, in the 70 degree flange contact plane.

Legend for line identification of following plots

————	solid :	leading outer wheel (or lead axle when only two lines are plotted)
-·-·-·-	single dot:	trailing outer wheel (or trailing axle when only two lines are plotted)
_ _ _ _	dashed :	leading inner wheel
- .. -	double dot:	trailing inner wheel

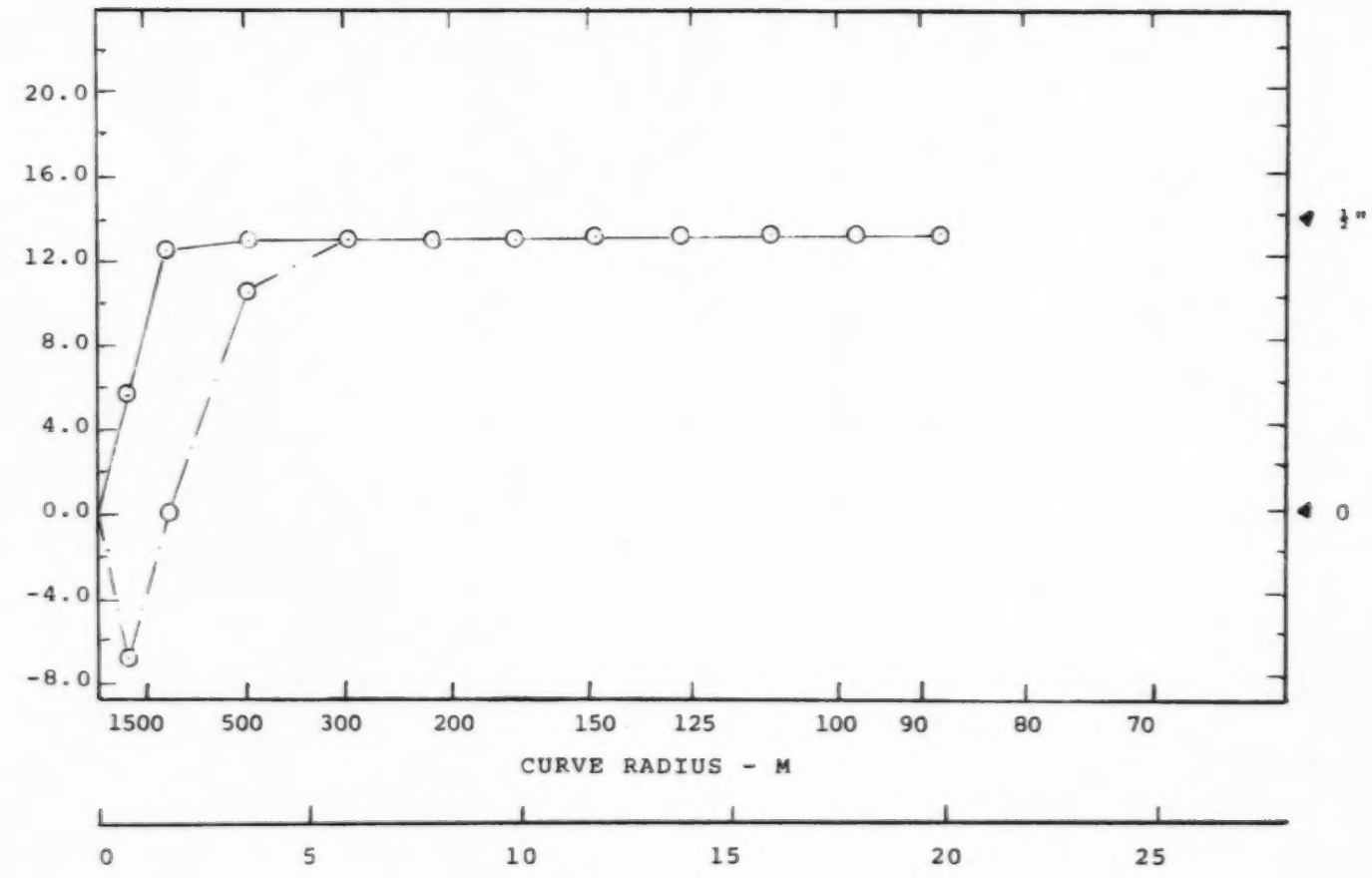
Lateral Tracking Position (mm)
20.0
16.0
12.0
8.0
4.0
0.0
-4.0
-8.0
½"
0
1500
500
300
200
150
125
100
90
80
70
CURVE RADIUS - M
0
5
10
15
20
25
TRACK CURVATURE - DEG

FIGURE 1

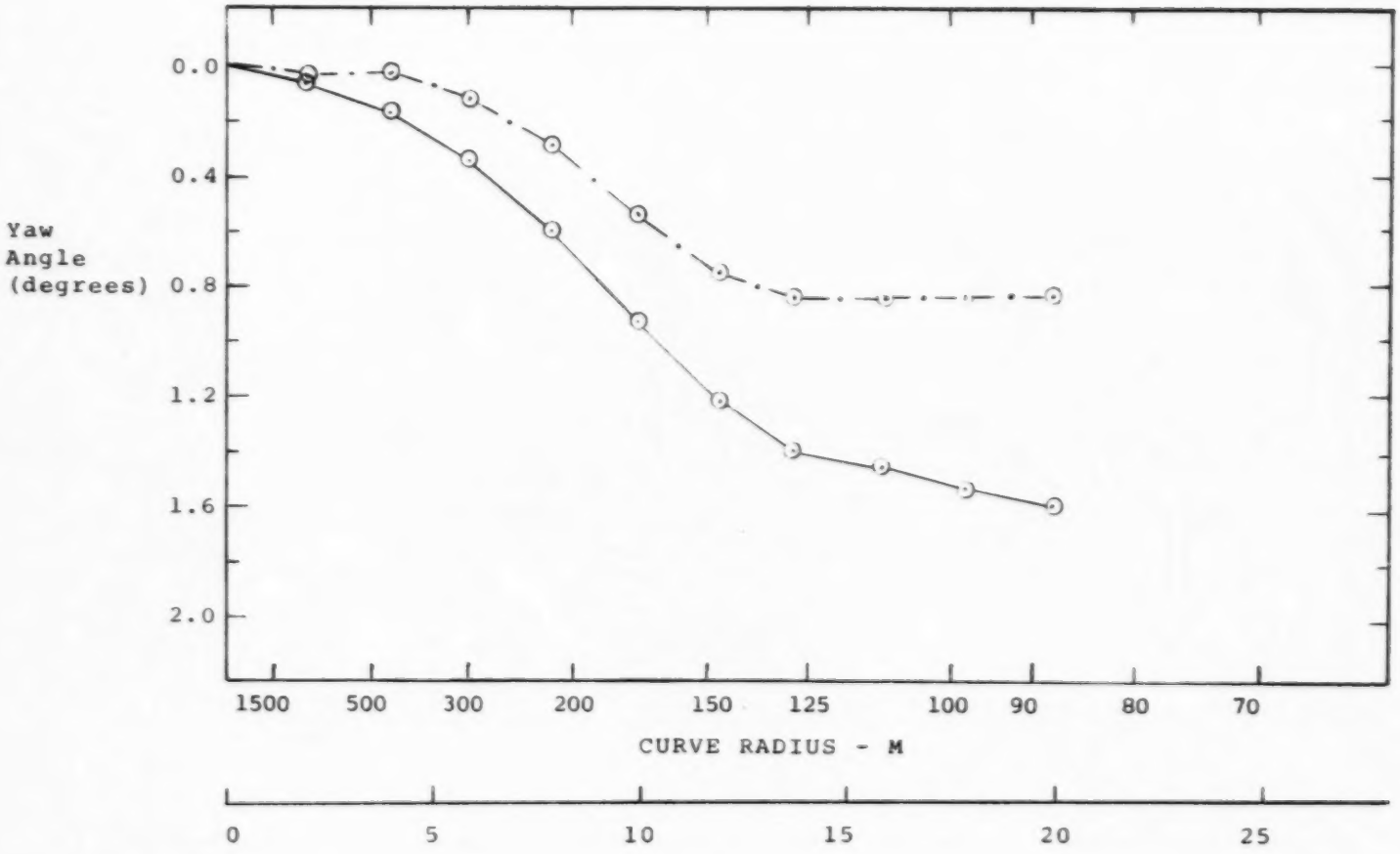
0.0
0.4
0.8
1.2
1.6
2.0
Yaw
Angle
(degrees)
1500
500
300
200
150
125
100
90
80
70
CURVE RADIUS - M
0
5
10
15
20
25
TRACK CURVATURE - DEG

FIGURE 2

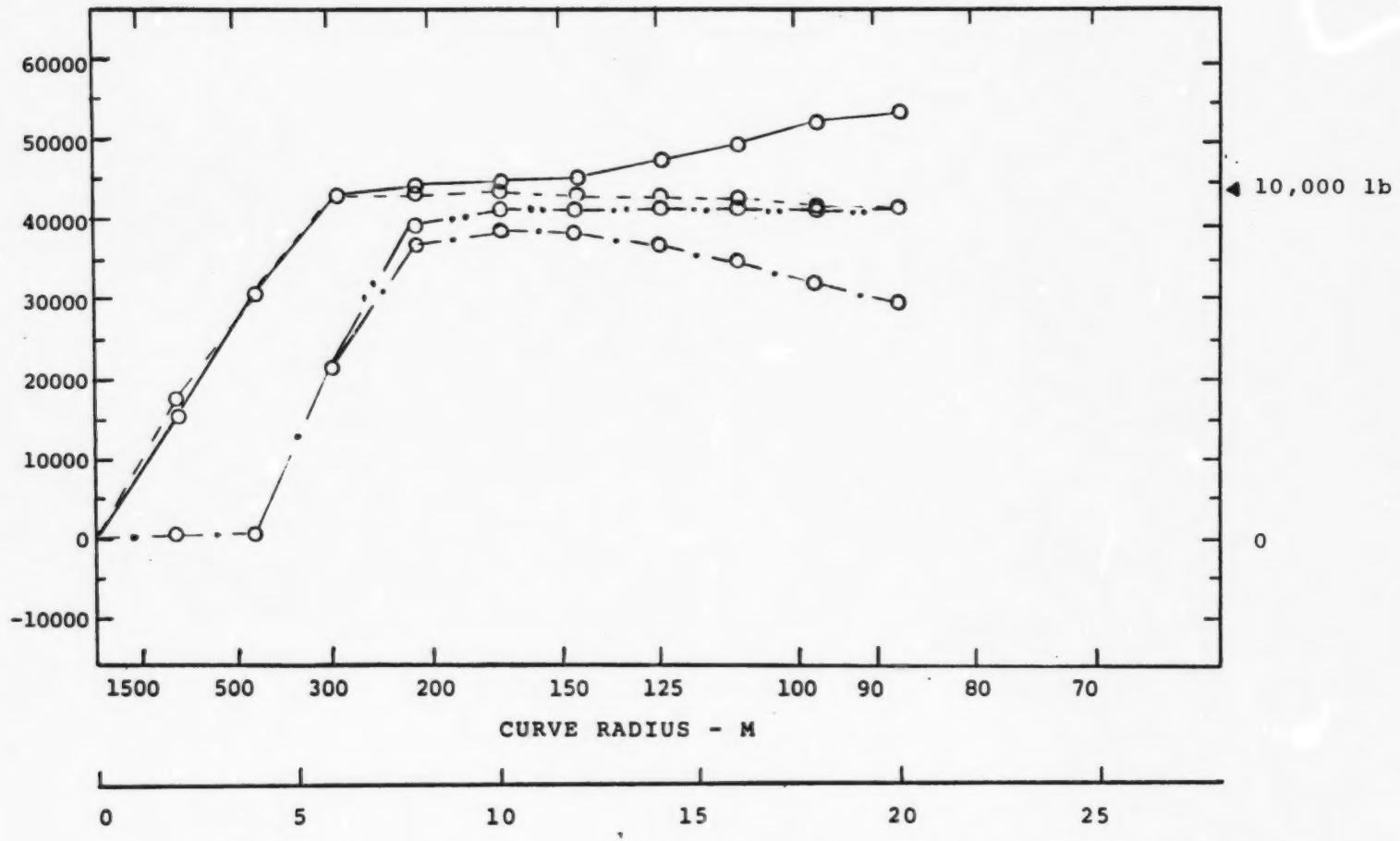
60000
50000
40000
30000
20000
10000
0
-10000
LATERAL
FORCE
N
10,000 lb
0
1500
500
300
200
150
125
100
90
80
70
CURVE RADIUS - M
0
5
10
15
20
25
TRACK CURVATURE - DEG

FIGURE 3

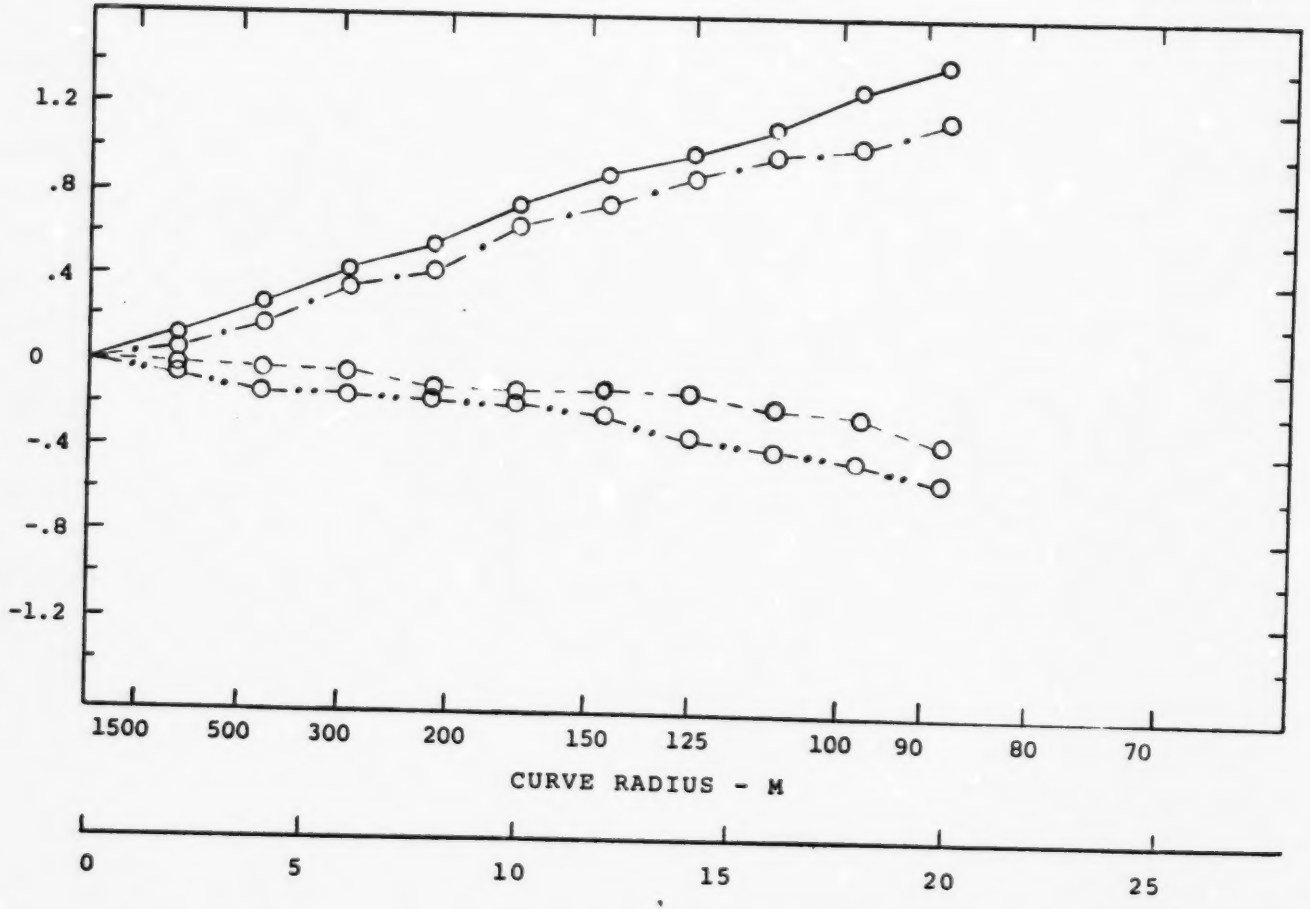
WHEEL
SLIP
(percent)
1.2
.8
.4
0
-.4
-.8
-1.2
1500
500
300
200
150
125
100
90
80
70
CURVE RADIUS - M
0
5
10
15
20
25
TRACK CURVATURE - DEG

FIGURE 4

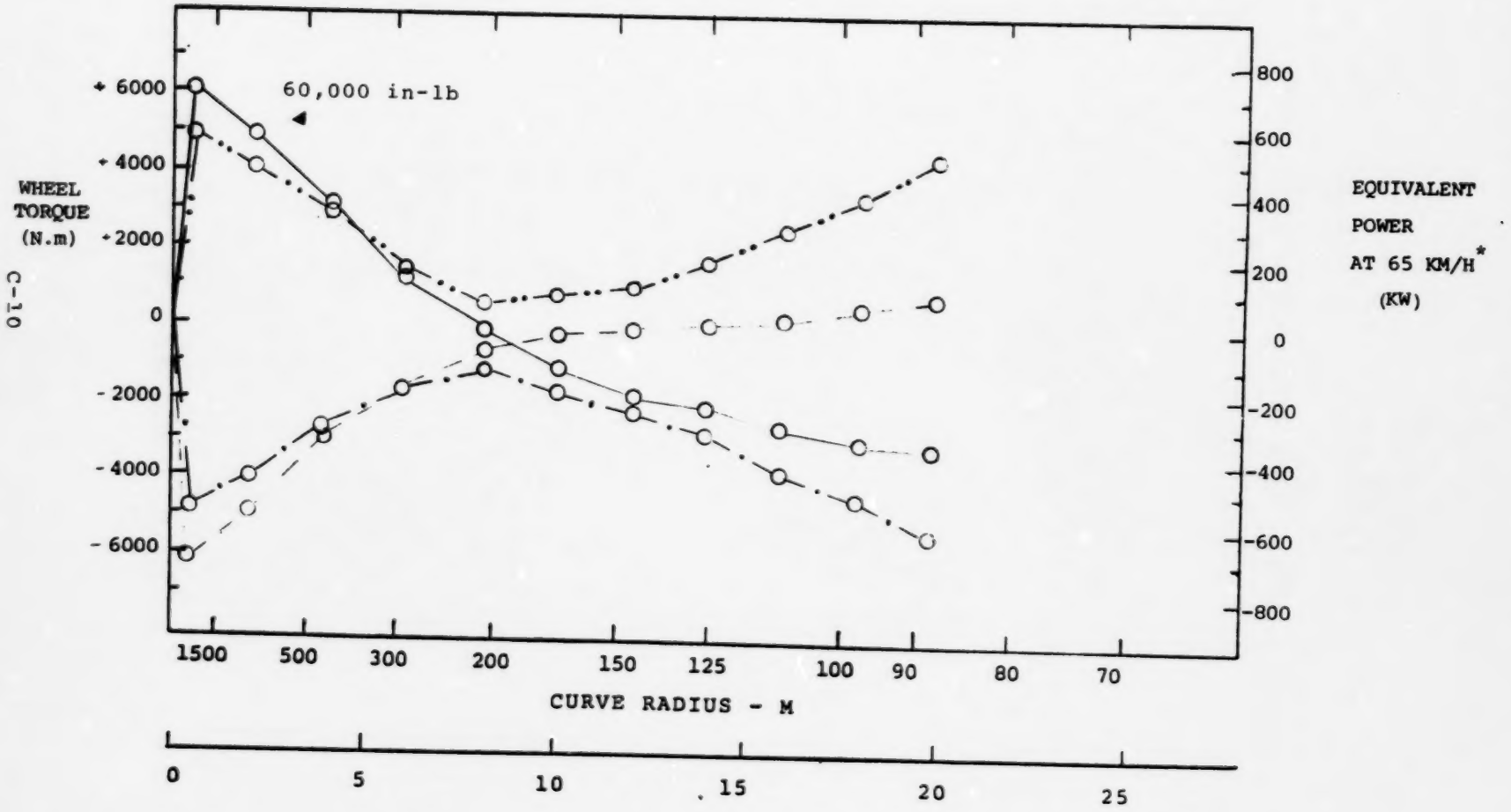

FIGURE 5

* Note: For illustrative purposes only. High Curvature not negotiable at this speed. For other speeds, adjust power proportionally to speed

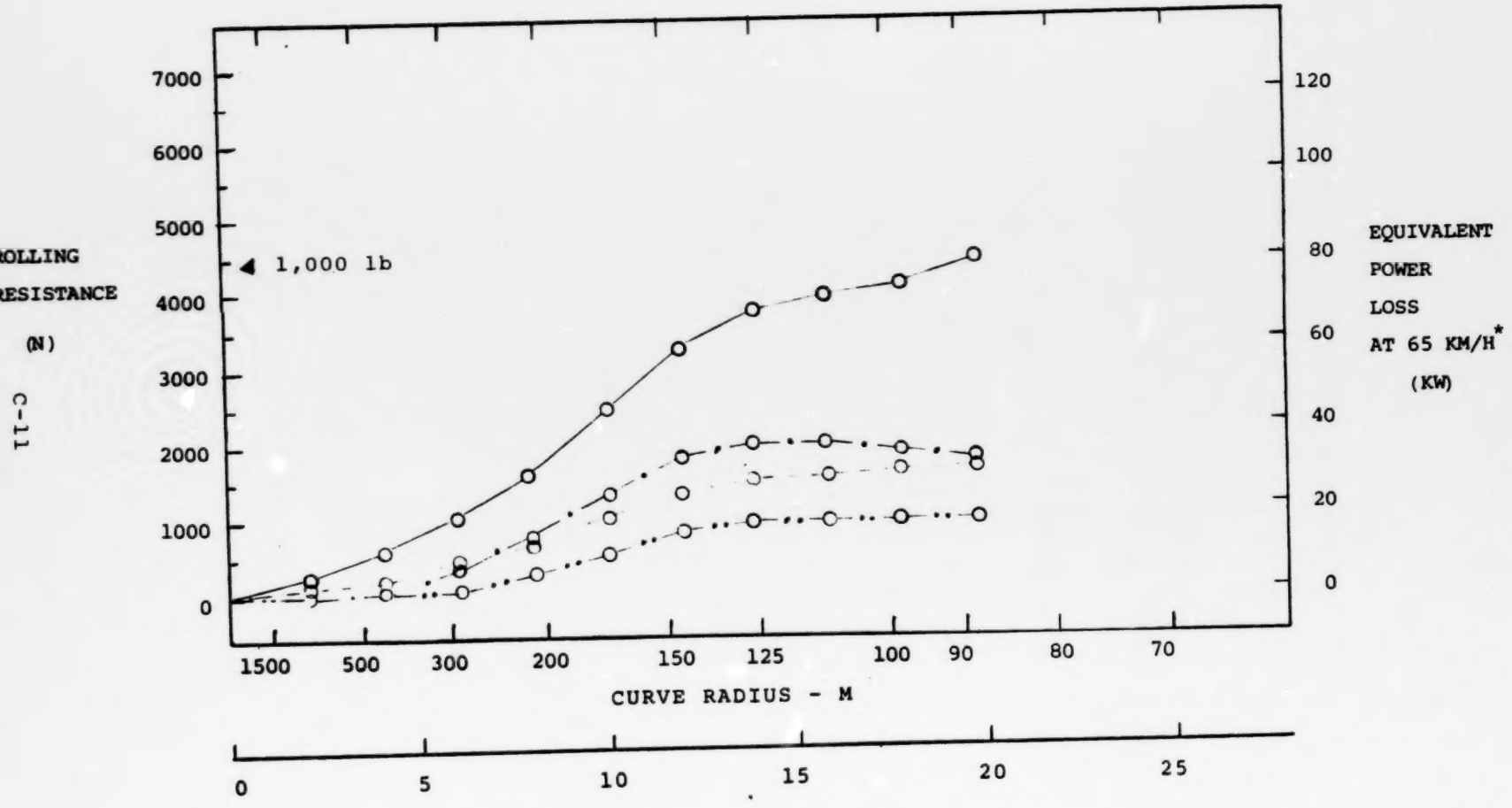

FIGURE 6

* Note: For illustrative purpose only. High Curvature not negotiable at this speed. For other speeds, adjust power proportionally to speed.

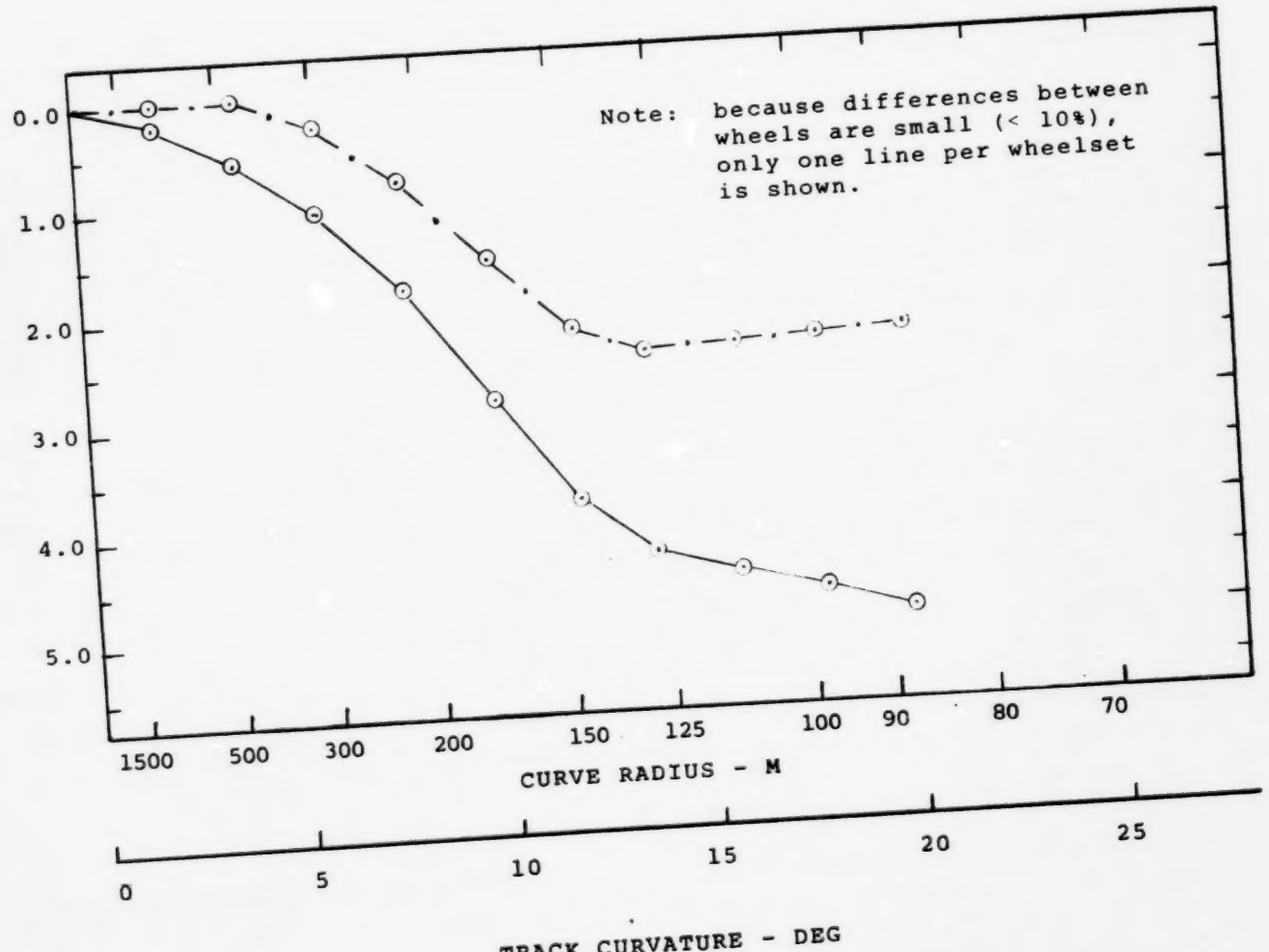
Note: because differences between wheels are small (< 10%), only one line per wheelset is shown.
TREAD CREEPAGE MAGNITUDE (percent)
0.0
1.0
2.0
3.0
4.0
5.0
1500
500
300
200
150
125
100
90
80
70
CURVE RADIUS - M
0
5
10
15
20
25
TRACK CURVATURE - DEG

FIGURE 7

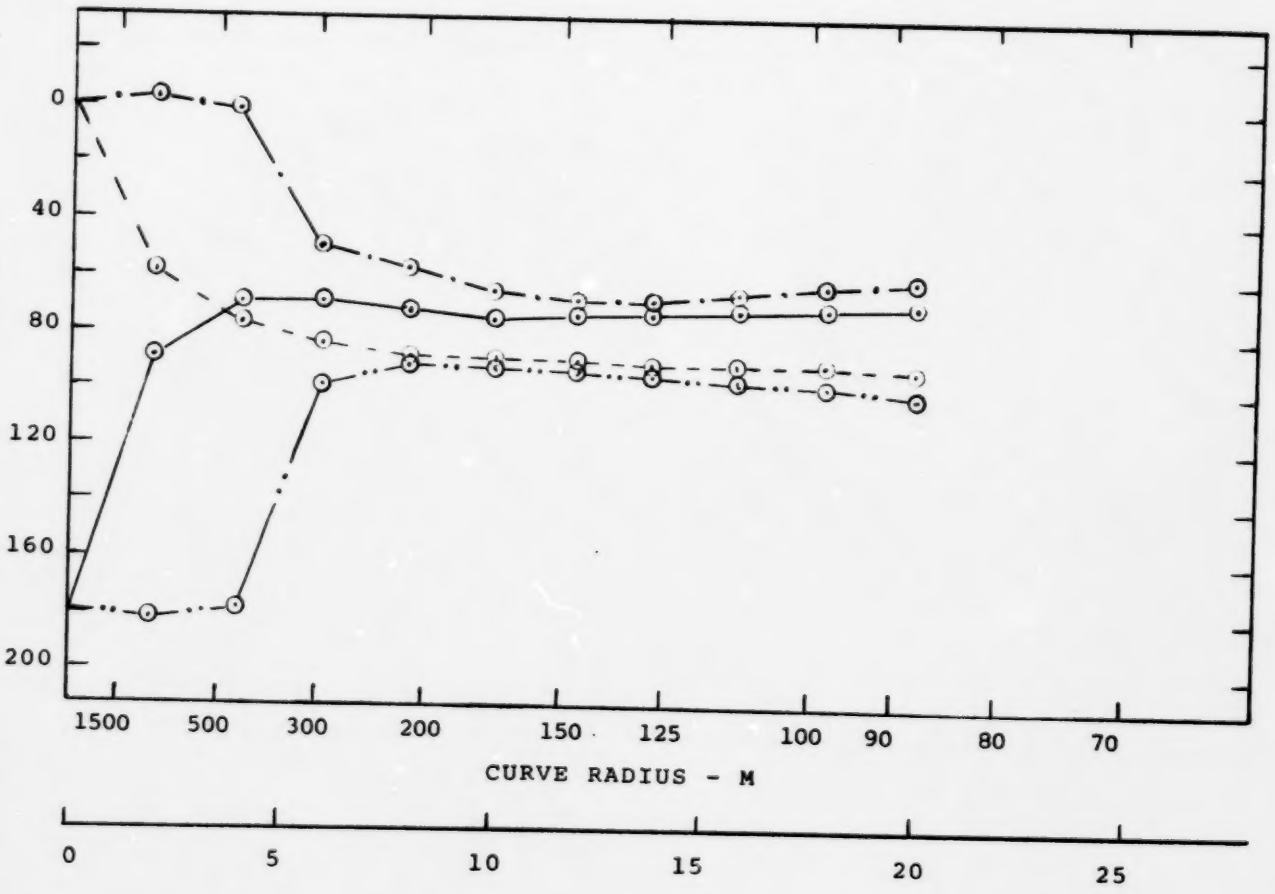
TREAD
CREEPAGE
DIRECTION
(degrees)
0
40
80
120
160
200
1500
500
300
200
150
125
100
90
80
70
CURVE RADIUS - M
0
5
10
15
20
25
TRACK CURVATURE - DEG

FIGURE 8

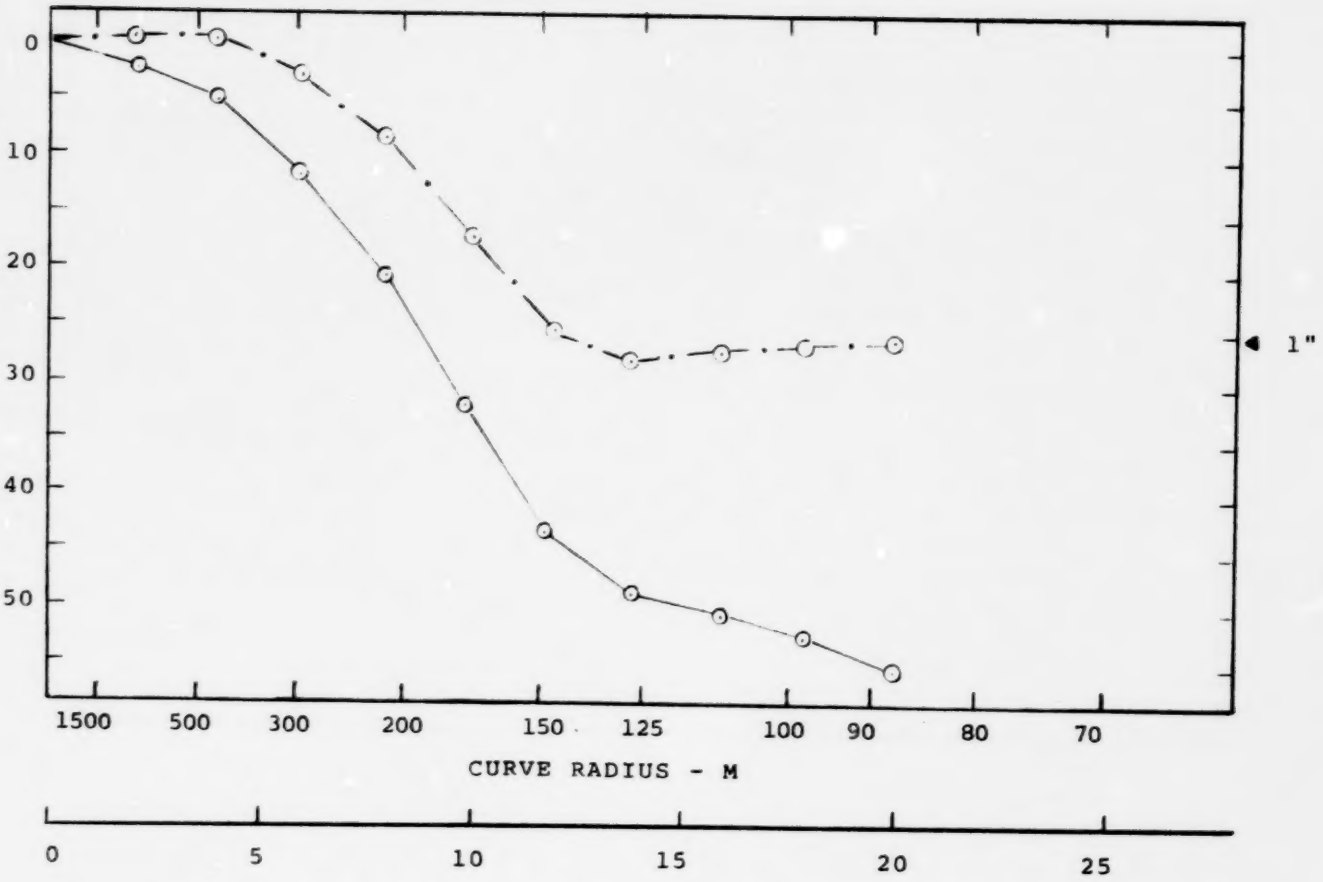
FLANGE CONTACT
POINT ADVANCE
(mm)
0
10
20
30
40
50
1"
1500
500
300
200
150
125
100
90
80
70
CURVE RADIUS - M
0
5
10
15
20
25
TRACK CURVATURE - DEG

FIGURE 9

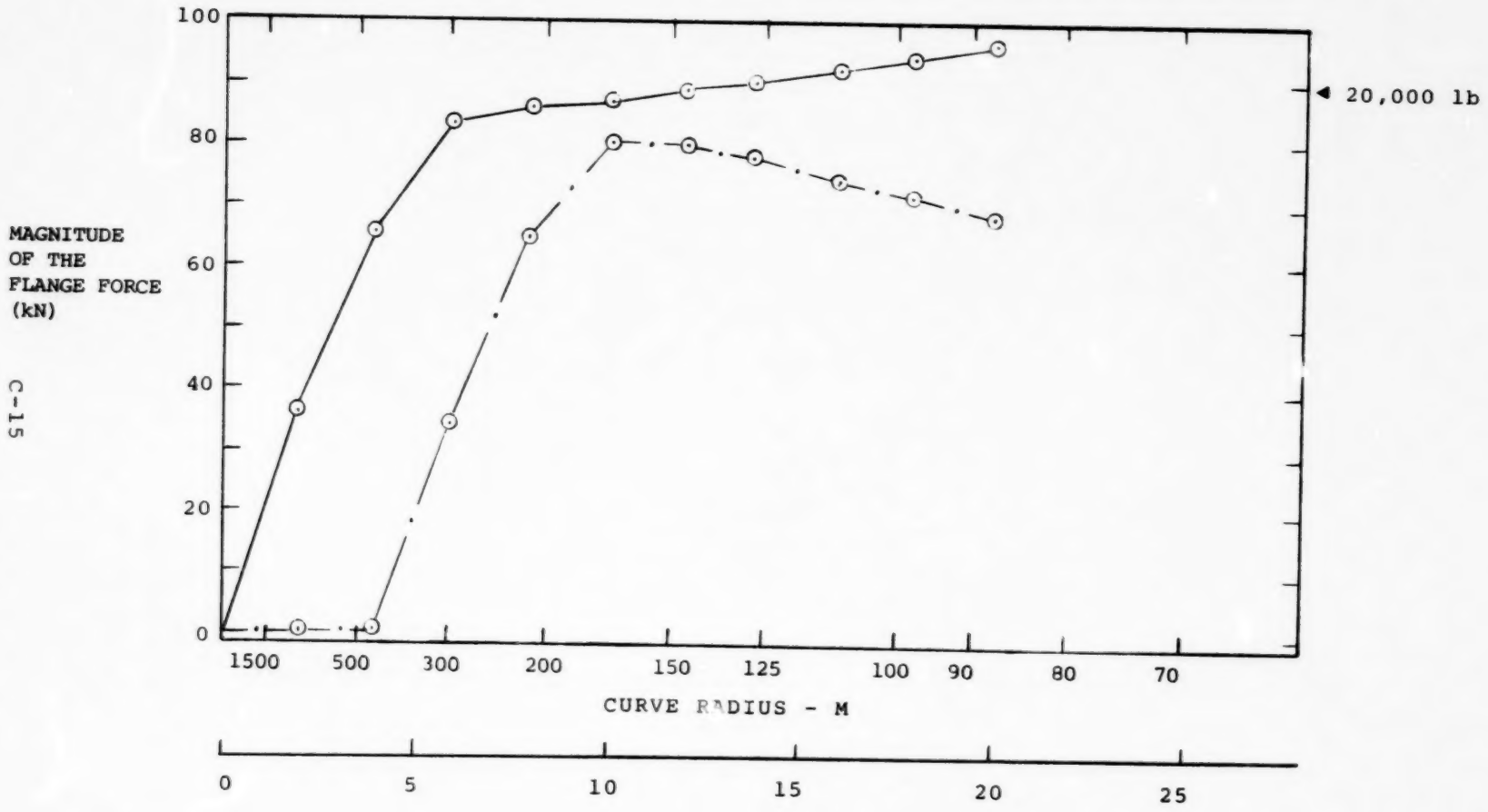
100
80
60
40
20
0
MAGNITUDE OF THE FLANGE FORCE (kN)
20,000 lb
1500
500
300
200
150
125
100
90
80
70
CURVE RADIUS - M
0
5
10
15
20
25
TRACK CURVATURE - DEG

FIGURE 10

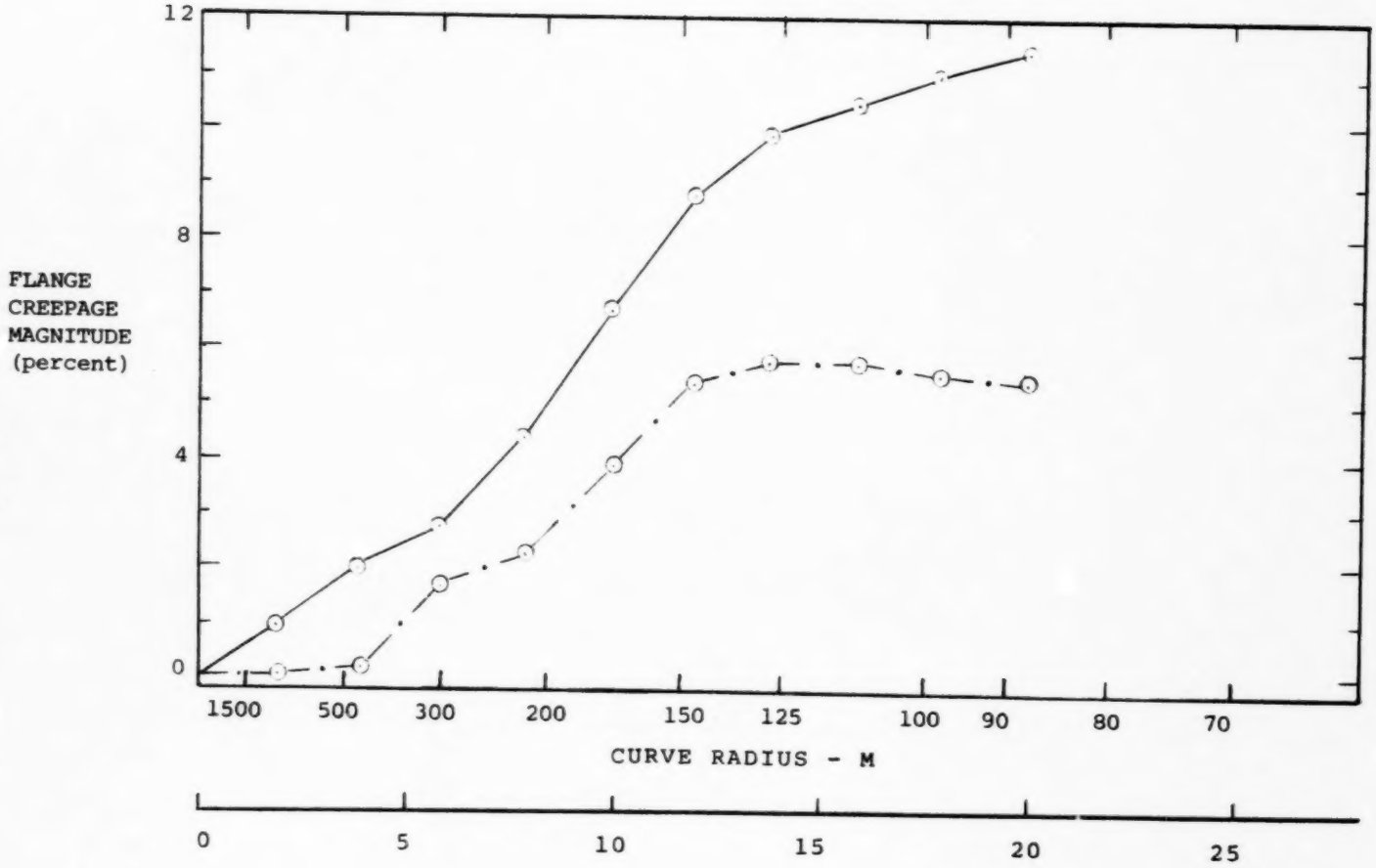
12
8
4
0
FLANGE
CREEPAGE
MAGNITUDE
(percent)
1500
500
300
200
150
125
100
90
80
70
CURVE RADIUS - M
0
5
10
15
20
25
TRACK CURVATURE - DEG

FIGURE 11

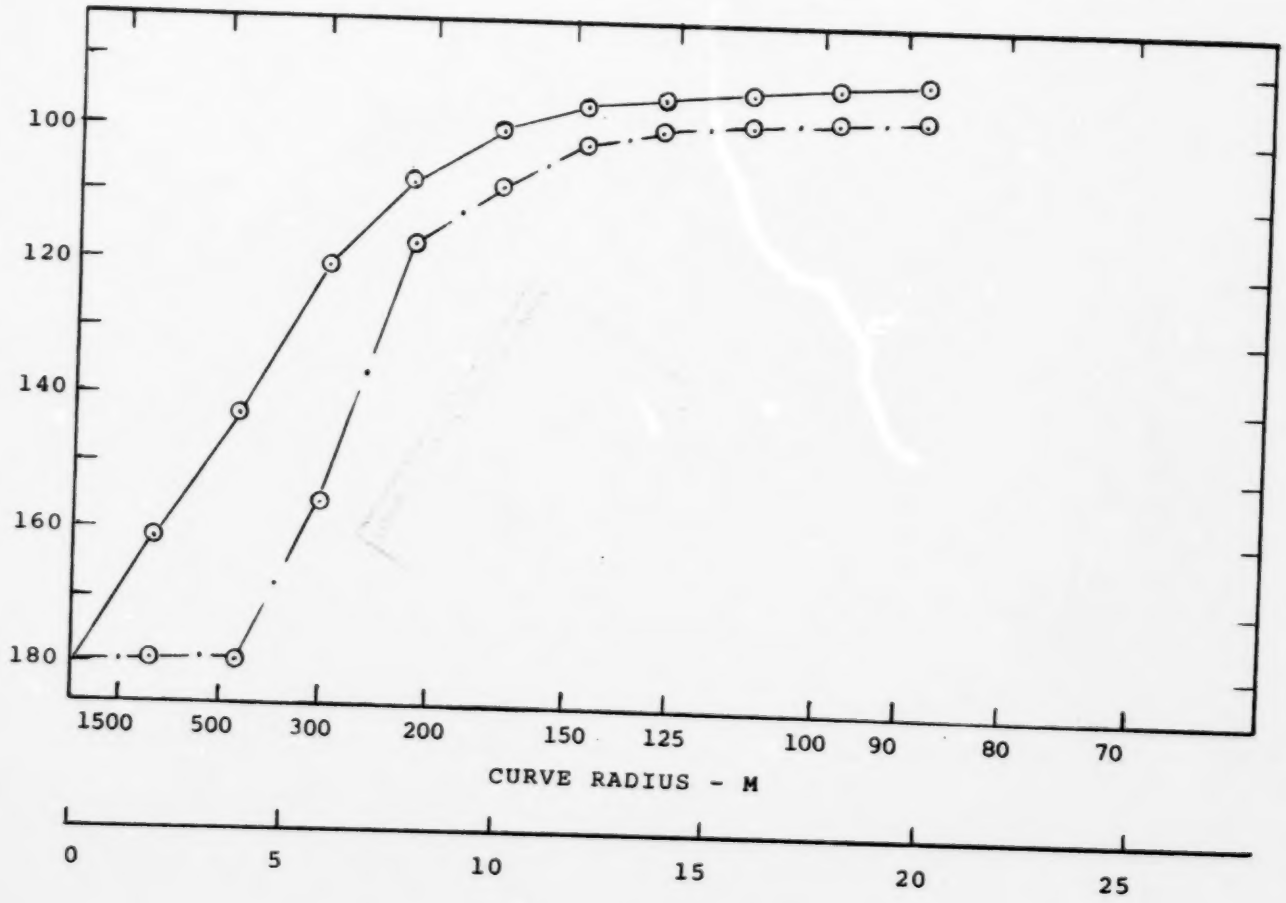
FLANGE
CREEPAGE
DIRECTION
(degrees)
100
120
140
160
180
1500
500
300
200
150
125
100
90
80
70
CURVE RADIUS - M
0
5
10
15
20
25
TRACK CURVATURE - DEG

FIGURE 12